



# Heat recovery from a cement plant with a Marnoch Heat Engine

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## ABSTRACT

This paper examines the performance of a new Marnoch Heat Engine (MHE) that recovers waste heat from within a typical cement plant. Two MHE units with compressed air as the working fluid are installed to recover the waste heat. The first unit on the main stack has four pairs of shell and tube heat exchangers. The second heat recovery unit is installed on a clinker quenching system. This unit operates with three pairs of shell and tube heat exchangers. The recovered heat is converted to electricity through the MHE system and used internally within the cement plant. A predictive model and results are presented and discussed. The results show the promising performance of the MHE's capabilities for efficient generation of electricity from waste heat sources in a cement plant. The new heat recovery system increases the efficiency of the cement plant and lowers the CO<sub>2</sub> emissions from the clinker production process. Moreover, it reduces the amount of waste heat to the environment and lowers the temperature of the exhaust gases.

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## 1. Introduction

The industrial processes of cement production generate a significant amount of CO<sub>2</sub> emissions. The first main source is from chemical reactions that occur during the calcination process:  $\text{CaCO}_3 + \text{Heat} \rightarrow \text{CaO} + \text{CO}_2$ . The second source of CO<sub>2</sub> emissions is the result of the combustion of coal, natural gas, or biofuels. A cement production process can be divided into three main steps: 1) preparing raw materials; 2) pyroprocessing in the clinker; and 3) grinding and blending in the clinker with other materials to produce different types of cement. The focus of this paper is heat recovery in the second step, which is the most energy intensive step.

Waste heat recovery in industrial processes can help to reduce the energy consumption and lower the greenhouse gas emissions. Waste heat can be captured from combustion exhaust gases, heated products, or heat losses from system processes [1]. Using waste heat to preheat the air for combustion affects the flame temperature and correspondingly NO<sub>x</sub> emissions [2]. In a typical cement plant, 25% of the total energy used is electricity, 75% is thermal energy, and about 35–40% of the process heat is lost by waste heat streams [3,4]. Approximately 26% of the heat input to the system is lost by dust, clinker discharge, radiation from the kiln and pre-heater surfaces, and convection from the kiln and pre-heaters [4]. A possible method to recover heat from within a cement plant is an

organic Rankine cycle [5]. This cycle can convert waste heat at high temperatures (598 K) to mechanical work [6].

The annual capacity of the Canadian cement industry is about 14.5 million tonnes [7]. About 5% of the total CO<sub>2</sub> emissions occurs from the cement industry [8]. From 1990 to 2004, CO<sub>2</sub> emissions from cement production in Canada increased by 31% due to the increase in clinker production capacity. This data indicates the importance of reducing the CO<sub>2</sub> emissions from the cement industry. Cement is produced by two main processes: a wet process and a dry process. In Canada, more than 80% of the cement is produced using the dry process and a precalciner/pre-heater type. Currently a large number of Canadian cement manufacturers use natural gas to supply heat for the rotary kiln. However, due to price fluctuations of natural gas, the cement industry has increasingly switched to coal as a substitute for the kiln fuel [7].

Heat engines can recover heat from various sources to produce mechanical work and hence electricity. They can be used to recover waste heat from cement plants. A heat engine converts thermal energy to mechanical energy, using temperature differentials. The term heat engine is often used in a broader sense to include work-producing devices that operate in a thermodynamic cycle. Daley [9] reported that hydrogen, as a working fluid, permits a more compact heat engine design than other working fluids (for a given power output and efficiency). Daley [9] reported that helium or air can be used as a gaseous working fluid when size is not a crucial factor. Due to the molecular weight and size of hydrogen molecules, those systems that use hydrogen as a working fluid must be well contained. Hydrogen molecules may penetrate through containment vessels [10]. Air as a working fluid has certain limitations, such as

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**Nomenclature**

|            |                          |
|------------|--------------------------|
| $\dot{E}x$ | exergy flow rate [W]     |
| $ex$       | specific exergy [J/kg]   |
| $h$        | specific enthalpy [J/kg] |
| $H$        | enthalpy [J]             |
| $\dot{m}$  | mass flow rate [kg/s]    |
| $\dot{Q}$  | heat flow rate [W]       |
| $T$        | temperature [K]          |
| $V$        | volume [m <sup>3</sup> ] |
| $\dot{W}$  | work rate [W]            |

*Greek Letters*

|          |                              |
|----------|------------------------------|
| $\Gamma$ | exergetic temperature factor |
| $\Delta$ | change                       |
| $\eta$   | efficiency                   |

*Subscripts*

|           |                                |
|-----------|--------------------------------|
| 0         | dead state (ambient condition) |
| 1, 2, ... | state 1, 2, ...                |

|        |                           |
|--------|---------------------------|
| A/F    | air/fuel ratio            |
| dest   | destroyed                 |
| e      | electric                  |
| Energy | energy                    |
| Ex     | exergy                    |
| F      | final                     |
| HHE    | hot heat exchanger        |
| HHEi   | hot heat exchanger inlet  |
| HHEo   | hot heat exchanger outlet |
| Hs     | heat source               |
| I      | initial, point number     |
| in     | inlet                     |
| loss   | loss                      |
| out    | outlet                    |
| Q      | heat                      |
| SI     | stack inlet               |
| SO     | stack outlet              |
| ta     | to air                    |
| w      | water                     |
| W/G    | water/glycol mixture      |

lower heat conductivity than hydrogen [11]. However, air as a working fluid is more convenient in systems because it is freely available and not reactive.

This paper develops and analyzes two new MHE (Marnoch Heat Engine; [12]) units to recover waste heat from a typical cement plant. This can significantly reduce the fuel consumption of the plant. The heat engine uses the available heat streams at a typical cement production site to produce electricity. Depending on the magnitude of the available waste heat streams at a site, an MHE with different capacities can be operated to improve the plant efficiency.

## 2. System description

The MHE operates similarly to other heat engines by converting thermal energy to mechanical work. This work is converted to electricity through an electric generator. The MHE consists of cylindrical shell and tube heat exchangers connected to a mechanical assembly to convert mechanical energy to electricity (see Fig. 1 and Table 1). Each heat exchanger is connected to a hot source and a cold sink via tubes and valves that allow a controlled mass flow at any step of the process. The MHE has isothermal expansions and isovolumetric heat additions similar to the Stirling cycle. However, there are three major differences between the MHE and a Stirling engine. In the MHE, the heat exchangers can be located apart from the piston assembly. Secondly, in the MHE, the enclosed area in the P–V diagram is reduced after each stroke as opposed to Stirling engines, in which the enclosed area remains almost constant. Fig. 2 demonstrates the second difference. In this figure, five sequential strokes of the MHE in one operation cycle for one pair of heat exchangers are shown. The enclosed area in the P–V diagram decreases after each stroke. Thirdly, in contrast to Stirling engines, the MHE has no regenerator in its mechanical configuration. All of these differences lead to advantages of the MHE over a Stirling cycle.

The MHE performs a conversion of thermal energy to flow work between two pressure vessels, then a subsequent conversion to mechanical energy by means of a specialized piston assembly, and finally a conversion to electricity by means of an electric generator. The engine consists of a pneumatic rotary actuator and a transmission system that converts mechanical energy provided through a flywheel to electricity in an electrical generator. Any cylinder configuration can be used and it is not necessarily restricted to

a rotary actuator. The transmission can be a belt drive or direct drive, similar to systems used in wind turbines. The differential in pressure between the heated tank and the cooled tank drives the actuator. The size of the actuator will depend on the size of the generator to be used. When the gas is highly compressed within the piston cylinder, the temperature differential needed to generate a sufficient pressure change is proportionately less than at a lower initial pressure.

In the proposed heat recovery system, two separate MHEs are used to capture waste heat from two different heat streams of a cement plant (see Fig. 3). The first MHE unit recovers heat from the main stack of the plant. The stainless steel coils of the hot heat exchangers are located inside the stack. These coils recover and transfer heat to the liquid in the first unit of the MHE. The heat exchangers that supply heat for the second MHE unit are located on the quenching system of the clinkers. The recovered heat from the clinkers is transferred through the second MHE unit. The secondary heat exchangers of the heat recovery system have shell and tube heat exchangers. As shown in Fig. 4, the heat exchangers are multi pass and multi inlet/outlet.

The minimum outlet gas temperature from the main stack in a clinker production system without a heat recovery unit is about 453 K. When gas flows through the coils of heat exchangers, the outlet temperature of the gas decreases by a maximum of 2 K. At a temperature of 451 K, no significant condensation of corrosive gases occurs inside the stack [1,13]. This prevents any potential damage and corrosion on the coils and stack surfaces.

The second unit of the MHE recovers quenching heat from the clinkers. In this section of the cement plant, heat can be recovered at lower temperatures. The temperature drop in this section is not problematic, since there is no corrosive gas in the quenching system. All of the coils of the main heat exchangers in the heat recovery system are made of stainless steel, so that heat, particles and corrosive gases would not cause significant problems.

## 3. Thermodynamic analysis of cement plant and Marnoch Heat Engine

### 3.1. Assumptions

The combustion heat comprises more than 95% of the required heat in the system. The amount of required thermal energy for

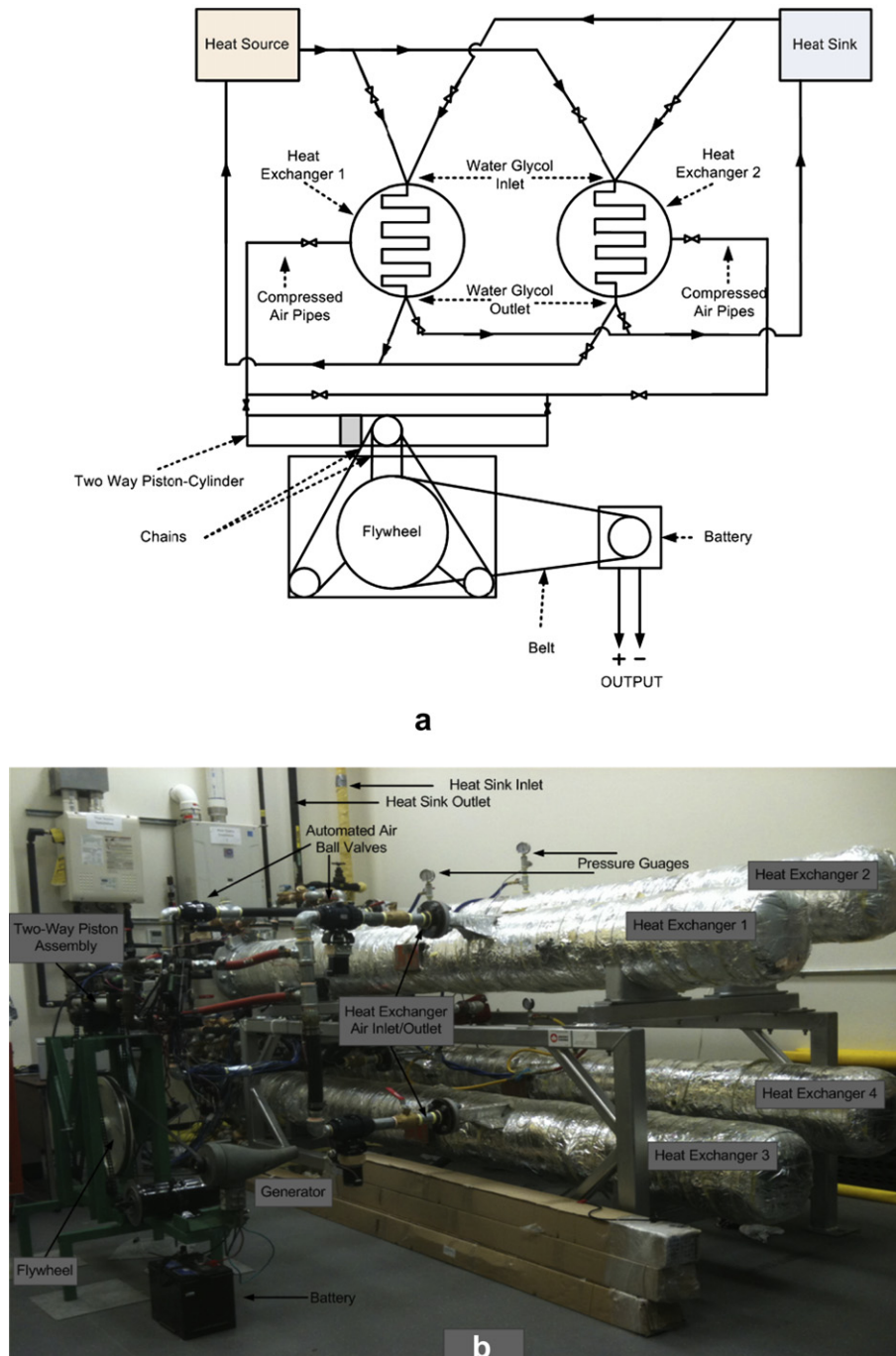


Fig. 1. a) Mechanical configuration of the MHE with two heat exchangers [12] and b) photo of MHE.

**Table 1**  
Heat exchanger specifications.

|                                              |            |
|----------------------------------------------|------------|
| Diameter of shell                            | 304.8 mm   |
| Length of shell                              | 3525.5 mm  |
| Number of baffles                            | 2          |
| Number of passes                             | 64         |
| Length of each pass                          | 3048 mm    |
| Diameter of tubes                            | 19.1 mm    |
| Total length of tubes                        | 195,072 mm |
| Percentage of occupied shell volume by tubes | 21.6%      |

clinker production is about 30.6 MJ/kg-clinker [4]. This can be supplied, for example, by about 0.115 kg of coal per kilogram of clinker. Equivalently, this energy can be extracted from 0.8 m<sup>3</sup>/kg-clinker of natural gas [14]. To analyze the thermodynamic system, the following assumptions will be adopted. The temperature and flow rate of the exhaust gases from the stack and the clinker cooling system are assumed to be constant. In this study, copper coils exposed to the ambient air are used as the heat sink. Axial belt driven fans are installed to move ambient air to the cooling coils, in case the removed heat from the cooling liquid as the result of natural convection is not sufficient for the heat engine. Also, air leakages from pipes, joints, and valves will be neglected. The

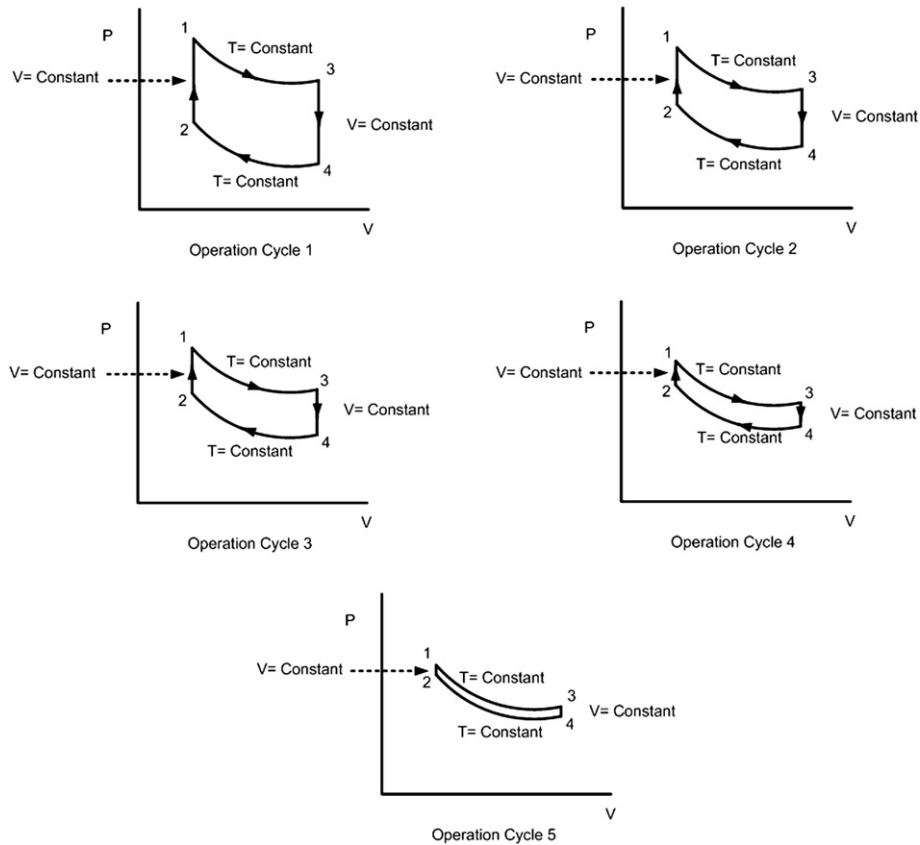


Fig. 2. Five sequential operation cycles of the MHE.

temperature and flow rate of the water moving through the heat exchangers will be assumed constant. An average ambient temperature of 288 K is assumed for the heat sink. Compressed dry air is the working fluid inside of the heat exchangers. Changes in the shell volume due to changes in the heat exchanger's temperatures and pressures will be neglected.

### 3.2. Energy and exergy analysis

Heat is recovered from two different sources. The first source is the kiln exhaust. The second source is hot air from the cooler that flows through the quenching system. The main coils of the second unit will capture heat from the air before it moves to the cyclones

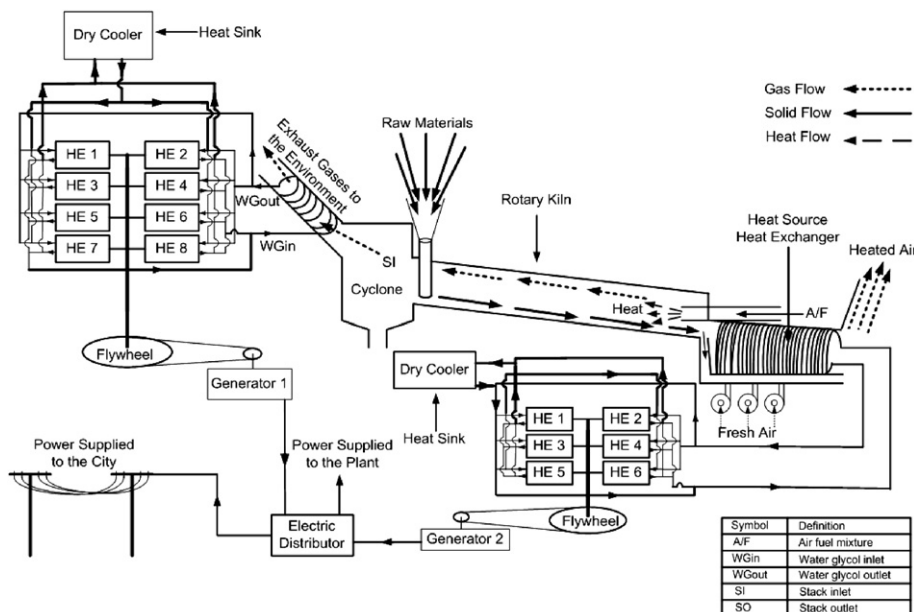


Fig. 3. Schematic of heat recovery system from a cement plant with two MHE units.

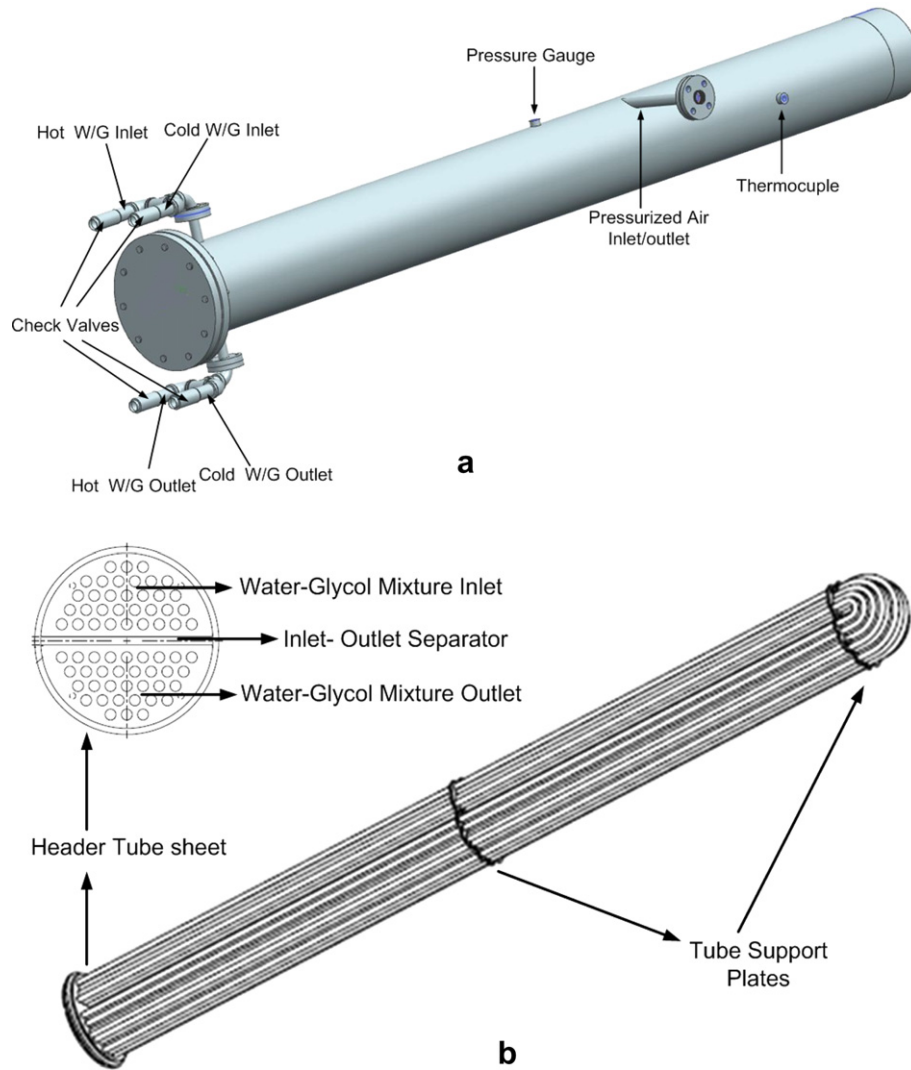


Fig. 4. Heat exchanger of the MHE: (a) Shell with liquid and gas inlets and outlets; (b) multi pass - multi inlet/outlet heat exchanger.

and bag-house. The hot stream from the kiln exhaust carries about 19% and the second stream from the cooler carries about 6% of the total system heat output. Engin and Vedat [4] have shown that a kiln with a capacity of 600 tons-clinker/day has a heat loss of about 4547 GJ/day via the two heat streams mentioned above.

### 3.3. Energy and exergy equations for the main stack

The exhaust gas that leaves the stack raises the temperature of the liquid. Equation (1) gives the mass balance for the air/fuel entering the system.

$$\dot{m}_{A/F} = \dot{m}_{SO} \quad (1)$$

A water/glycol mixture flows through the tubes of the heat exchangers in the stack. When the mixture leaves the stack, it flows through the hot heat exchangers of the MHE.

$$\dot{m}_{WGIn} = \dot{m}_{WGout} \quad (2)$$

The fluid inside of the helical tubes of the stack absorbs heat from the exhaust gases and supplies heat to the engine for gas expansion. Equation (3) shows the amount of heat gained from the stack under steady state operation in the stack heat exchanger.

$$\dot{m}_{A/F} h_{SI} + \dot{m}_{WGIn} h_{WGIn} = \dot{m}_{SO} h_{SO} + \dot{m}_{WGout} h_{WGout} + \dot{Q}_{loss} \quad (3)$$

The ambient temperature and pressure are assumed to be 288 K and 103.4 kPa, respectively. These conditions correspond to the dead state for the exergy analysis. Equation (4) shows the exergy balance equation for the main stack.

$$\dot{m}_{A/F} ex_{SI} + \dot{m}_{WGIn} ex_{WGIn} = \dot{m}_{SO} ex_{SO} + \dot{m}_{WGout} ex_{WGout} + \dot{E}_{x_{loss}} + \dot{E}_{x_{dest}} \quad (4)$$

where  $\dot{E}_{x_{loss}}$  is the exergy loss of the heat loss from the heating system to the environment.  $\dot{E}_{x_{dest}}$  is the amount of exergy destroyed through the system. The exergy may be destroyed due to friction losses or heat transfer to/from the fluid flow through the tube bundles. In the next section, the physical exergy ( $ex$ ) and exergy associated with heat transfer will be examined.

### 3.4. Exergy analysis

To determine the performance of the system in different ambient conditions, an exergy analysis will be performed. Exergy is the measure of energy availability [15,16]. Exergy analysis deals with irreversibility within the system and it is based on the second law



efficiency [17,18]. Exergy analysis is a useful tool to design and improve the system [19–21]. In the MHE, no chemical reaction occurs with the air or the water/glycol, which transfers heat from the system.

The physical exergy of each point  $Ex_i$  can be calculated by:

$$Ex_i = \dot{m}[(h_i - h_0) - T_0(s_i - s_0)] \quad (5)$$

where  $h_0$ ,  $T_0$  and  $S_0$  are specific enthalpy, temperature and entropy of the dry air under ambient conditions, respectively. Also,  $h_1$  and  $S_1$  are the specific enthalpy and entropy at each point within the system. To find the exergy flow,  $\dot{Ex}_Q$ , which is associated with heat transfer into the system,  $\dot{Q}$ , Equation (6) can be used. Also, heat losses from the shells and coils are calculated as follows,

$$\dot{Ex}_Q = \int_I^F \left(1 - \frac{T_0}{T}\right) d\dot{Q} \quad (6)$$

where the integral is taken from the initial state ( $I$ ) to the final state ( $F$ ). This thermal exergy is the minimum required work to bring the control mass from the dead state to the final state. In the MHE units, in most conditions, heating starts from the dead state. In situations where the temperature of the heat sink is lower than the ambient temperature, the initial state has negative exergy values.

The exergetic temperature factor is defined as

$$\Gamma = \left(1 - \frac{T_0}{T}\right) \quad (7)$$

For heat transfer across a segment ( $r$ ) of the control surface, for which tanks 2, 4, 6 and 8 are initially charged, the exergy transferred into the first MHE unit is

$$\dot{Ex}_Q = \int_r q_r \Gamma dA_r \quad (8)$$

where  $q_r$  is the heat flow per unit area at a region on the control surface, and the temperature is denoted as  $T_r$ . The dimensions of the stainless steel tubes inside the heat exchanger are used to find the surface area.

Equation (9) can be used instead of Eq. (8) when the value of  $\dot{Q}_{ta}$  is known. If the temperature  $T$  of the control mass is constant, the thermal exergy transfer associated with the heat transfer is:

$$\dot{Ex}_Q = \left(1 - \frac{T_0}{T}\right) \dot{Q}_{ta} = \Gamma \dot{Q}_{ta} \quad (9)$$

where  $T_0$  is the ambient temperature and  $T$  is the working fluid temperature during the discharging mode.

### 3.5. Mass balance equations for the MHE heat exchangers

In this section, all of the mass balance equations, related to one operational cycle, for the shell and tube heat exchangers in the MHE are analyzed. In the first operation cycle, the hot water/glycol mixture flows to heat exchangers 2, 4, 6, and 8. Hence the working fluid (pressurised air) inside these heat exchangers reaches a higher pressure. The other four heat exchangers are cooled with the mixture that is flowing from the heat sink. The dry air inside these shells is de-pressurising.

The mass balance equation for the liquid in each hot heat exchanger of the MHE is:

$$\dot{m}_{WGin} = (1/4)\dot{m}_{HHE} \quad (10)$$

where  $\dot{m}_{WGin}$  is the total mass flow rate passing through the main stack and  $\dot{m}_{HHE}$  is the mass flow rate of the heated fluid through each hot heat exchanger (Fig. 5). Equation (10) shows that the water/glycol flow rate in each individual heat exchanger is in steady state. The liquid mass flow rate inside the steel coils in each heat exchanger is constant. It has a quarter value of the water/glycol that flows through the main stack coils. In the second MHE unit, the mass flow rate of the water/glycol through each individual heat exchanger is one third of the total flow rate through the main coils inside the quenching system. For the air, individual heat exchangers of the system are not in steady state.

$$\sum \dot{m}_{Air,in} - \sum \dot{m}_{Air,out} = \Delta \dot{m} \quad (11)$$

In each MHE operation cycle, each heat exchanger has three modes. As shown in Fig. 6, these modes are controlled by two automated ball valves. At first, increasing and decreasing the temperatures in each pair of heat exchangers causes pressure

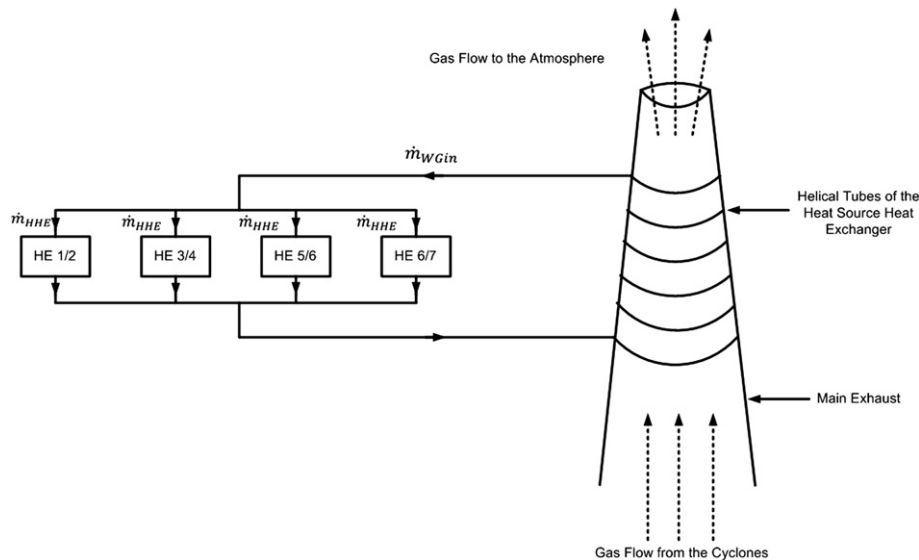


Fig. 5. Heat supply for the first heat recovery unit.

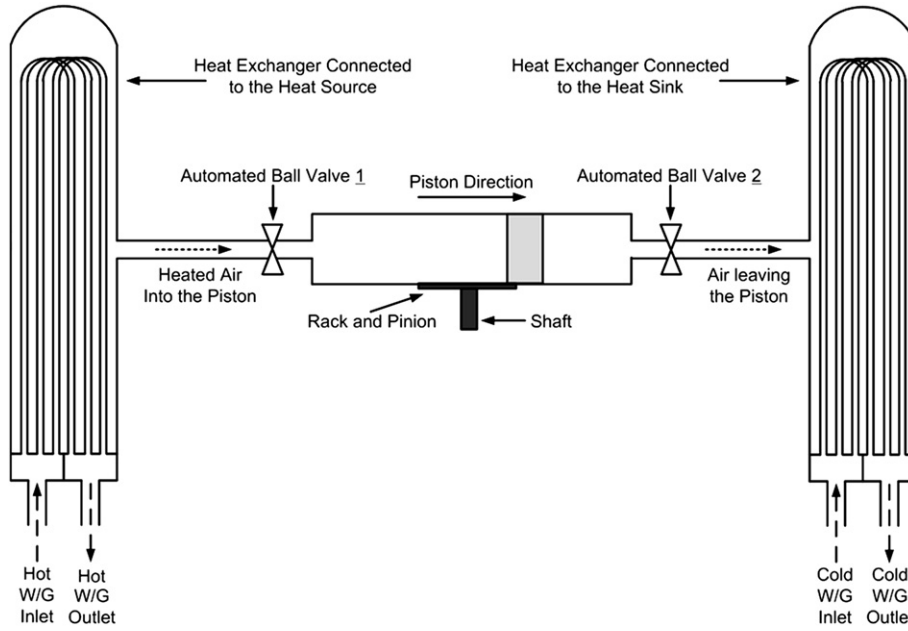


Fig. 6. Piston assembly connected to one pair of heat exchangers.

differences. During this period, no air mass transfer occurs between the heat exchangers. In the second mode, the air mass flow from the high pressure heat exchanger passes through the piston assembly and eventually flows to the low pressure side. To keep the process approximately isothermal, water flows through the heat exchangers simultaneously. When the air expands in the piston assembly, it flows towards lower temperature areas. Heat is required to sustain the gas temperature during the expansion process. If the amount of supplied heat to the working fluid is sufficient to keep the temperature constant, then isothermal expansion occurs. The same situation exists in the cold heat exchangers to attain isothermal compression. The amount of removed heat from the air must be equal to the generated heat from the compression process. Isothermal expansions and compressions result in extracting more cyclic work from the MHE heat engine. As mentioned earlier, air leakages from the valves and joints are assumed negligible. Thus, in Eq. (11), the parameters for heating and cooling modes for a set of heat exchangers are zero when air valves are closed (no air enters or leaves the shell).

At the start of the piston operation, air leaves the heated heat exchanger and no air mass enters the shell. For the period of  $dt$ , the mass flow rate leaving the piston assembly is

$$\Delta \dot{m} = -\frac{dm_{\text{Air,Out}}}{dt} \quad (12)$$

where  $m_{\text{Air,out}}$  is the air mass flow rate that is leaving the hot heat exchanger. After each piston stroke, the amount of air that is transferred from the heat source to the heat sink decreases. This results in decreasing the cyclic work output from the MHE unit (Fig. 4).

When the pressures of both shells are equal, no air mass will be transferred from the heat exchanger. At any instant of time during the process, the continuity equation is given by

$$\frac{m_{\text{Air,out}}}{dt} + \sum \dot{m}_{\text{HHE}} = 0 \quad (13)$$

where the summation is the overall area on the control surface across which the flow occurs. In shell 1, only the exit point area is

considered for Equation (13). Integrating over  $t$  gives the changes of mass in shell 1 during the overall process:

$$\int_0^t \left( \frac{m_{\text{Air,out}}}{dt} \right) dt = m_{\text{initial}} - m_{\text{final}} \quad (14)$$

where  $m_{\text{initial}}$  denotes the initial air mass in shell 1, and  $m_{\text{final}}$  denotes the air mass that remains in shell 1 after one operation cycle. Thus, the initial and final properties of the gas inside the MHE heat exchanger allow the value in Eq. (14) to be calculated. This equation will be solved for the whole period of the discharging process.

### 3.6. Energy and exergy analysis of the MHE heat exchangers

In the first cycle of operation, as shown in Figs. 3 and 5, heat exchanger 2 is connected to the heat reservoir. Thus, for the hot water stream passing through the tube bundle of the MHE heat exchangers, the energy balance equation is:

$$\dot{m}_{\text{HHE}} h_{\text{HHEi}} = \dot{m}_{\text{HHE}} h_{\text{HHEo}} + \dot{Q}_{\text{ta}} + \dot{Q}_{\text{loss}} \quad (15)$$

where  $\dot{m}_{\text{HHE}}$  is equal to the value from Equation (10), and  $h_{\text{HHEi}}$  and  $h_{\text{HHEo}}$  denote the specific enthalpy of the water/glycol that enters and exits the heat exchanger, respectively. This  $\dot{m}_{\text{HHE}} h_{\text{HHEi}}$  gives the energy flow rate that is entering the system by the water stream to one heat exchanger. The temperature of the water/glycol that is entering the system is assumed to be constant during the charging and operation modes. Hence, there will be no change in the value  $\dot{m}_{\text{HHE}} h_{\text{HHEi}}$  over time. Unlike  $\dot{m}_{\text{HHE}} h_{\text{HHEi}}$ ,  $\dot{m}_{\text{HHE}} h_{\text{HHEo}}$  changes over time and its value depends upon  $\dot{Q}_{\text{ta}}$ . This rate is reduced over the period of time during charging, as the temperature of the gas approaches the liquid temperature. Also,  $\dot{Q}_{\text{loss}}$  is the amount of heat rejected to the ambient from the hot stream tubes and it varies with time. In the MHE system, the hot water pipes into the tanks are well insulated. Thus,  $\dot{Q}_{\text{loss}}$  can be neglected.

The heat transfer rate ( $\dot{Q}_{\text{ta}}$ ) from the water/stream to the air, over a certain period of time is expressed as:

$$\int_0^t \dot{Q}_{ta}(dt) = mc_v \Delta T \quad (16)$$

When Eq. (16) is used for the charging period, the air mass ( $m$ ) is constant inside the shell, and its value is  $m_{\text{initial}}$  in Eq. (11). Temperature differences of  $\Delta T$  have major effects on the heat transfer rates. When this equation is solved for the water streams,  $\Delta T$  is the temperature difference between the water/glycol inlet and outlet.

When the piston operation starts, heat is transferred by convection through the piston assembly, and afterwards it is transferred into the cold heat exchanger. When the air mass is leaving the hot heat exchanger shell, the pressure of the shell will decrease. However, the expansion process in this heat engine should be near isothermal to obtain a maximum work output. Water flows through the heat exchangers during piston operation, so when the gas is expanded, it is simultaneously heated. The amount of required heat to keep the first expansion isothermal is not equal to the amount of required heat that maintains the temperature of the last expansion consistent. In the proposed system, the mass flow rate and temperature of the water/glycol that flows into the system is maintained constant. Hence there is a variation between the actual expansions and theoretical isothermal expansions.

The energy balance equation inside heat exchanger 2 in the first operation cycle, during the operation mode, can be expressed as:

$$\Delta \dot{E} = \dot{Q}_{ta} - \dot{m}_{\text{HHE}} h_{\text{HHEo}} \quad (17)$$

where  $\Delta \dot{E}$  is the transient change of state in heat exchanger 2 that occurs due to heat and mass transfer. The value of  $\dot{Q}_{ta}$  is obtained from Eq. (15),  $\dot{m}_{\text{HHE}}$  is calculated in Eq. (9), and  $h_{\text{HHEo}}$  is the specific enthalpy of the air at the outlet.

Equation (18) shows the exergy balance equation for one of the MHE heat exchangers when it is operating in the heating mode. The MHE heat exchangers have a transient operation, because the amount of air inside the shell changes after each piston stroke. Hence, the value of  $\dot{E}x_{ta}$  must be calculated after each stroke,

$$\dot{m}_{\text{HHE}} ex_{\text{HHEi}} = \dot{m}_{\text{HHE}} ex_{\text{HHEo}} + \dot{E}x_{ta} + \dot{E}x_{\text{loss}} + \dot{E}x_{\text{dest}} \quad (18)$$

where  $ex_{\text{HHEi}}$  is the exergy of the water/glycol mixture at the heat exchanger inlet. Also,  $ex_{\text{HHEo}}$  is the exergy of the mixture at the exit point, and  $\dot{E}x_{ta}$  is the exergy associated with heat that is transferred from the liquid to the air inside the heat exchanger. This value changes after each stroke because of the changes in air mass inside the shells.  $\dot{E}x_{\text{loss}}$  is the exergy loss associated with the heat flow. The exergy loss from the coils is assumed to be negligible.  $\dot{E}x_{\text{dest}}$  is the exergy destroyed through the tube bundle. Minimizing the exergy loss increases the available exergy transferred to the compressed air inside the shell, thereby leading to higher pressures inside the shells. In order to reduce the pressure drop inside the tubes, multi pass and multi entrance heat exchangers are used for the MHE unit (see Fig. 4).

### 3.7. MHE system efficiency

In order to find the overall energy efficiency of the MHE, the electricity output ( $\dot{W}_e$ ) from the engine was measured. This value is divided by the thermal energy supplied to the heat engine. The overall efficiency of the MHE is given as

$$\eta_E = \frac{\dot{W}_e}{\dot{Q}_{in}} \quad (19)$$

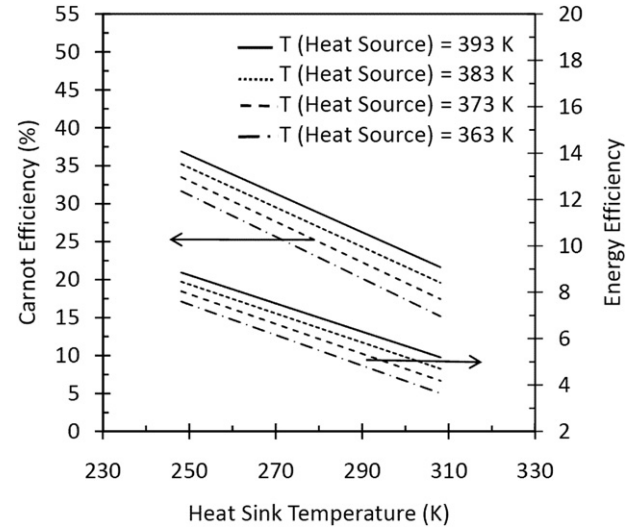


Fig. 7. Variation of ideal and energy efficiency at different heat sink temperatures.

where  $\dot{Q}_{in}$  is calculated by  $\dot{m}_{\text{W/Gin}} h_{\text{W/Gin}}$ . Higher W/G flow rates at higher temperatures lead to higher thermal energy input into the MHE.

The exergy efficiency of the system is given by

$$\eta_{\text{Ex}} = \frac{\dot{W}_e}{\dot{E}x_{in}} \quad (20)$$

where  $\dot{E}x_{in}$  is the exergy associated with heat transfer into the system. This value is calculated by multiplying the exergetic temperature factor  $\left(1 - \frac{T_0}{T}\right)$  by the heat transfer from the hot stream into the system, as shown in Equation (19) by  $\dot{Q}_{in}$ .

$$\dot{E}x_{in} = \left(1 - \frac{T_0}{T}\right) \dot{Q}_{in} \quad (21)$$

In the next section, sample results from the prior analysis will be presented and discussed.

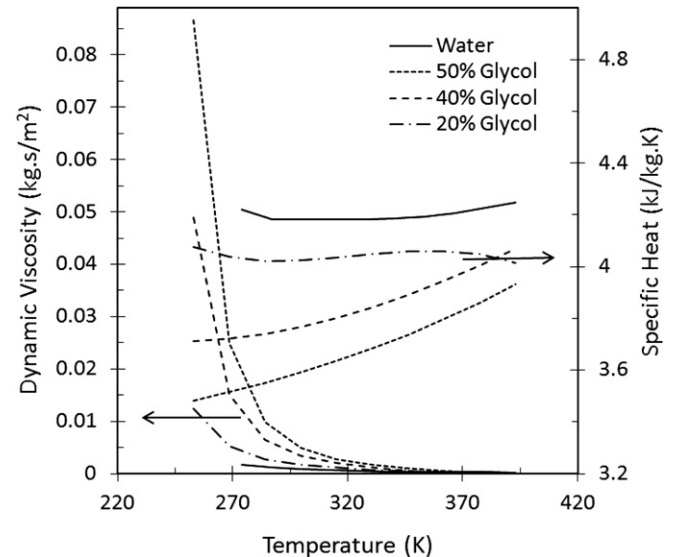


Fig. 8. Variation of specific heat capacity and dynamic viscosity of the water/glycol mixtures at different temperatures.



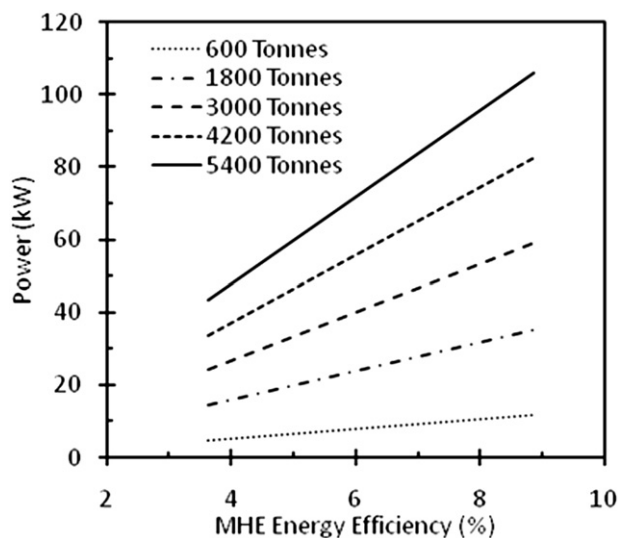


Fig. 9. Total power output from the generator at varying energy efficiencies.

#### 4. Results and discussion

Higher clinker production results in a larger gas flow rate through the stack and more heat available for recovery. Since more clinker production is available for quenching, the amount of available energy for the second MHE unit increases as well. The amount of heat recovered from the system varies through different seasons of the year because the ambient temperature changes, and therefore the MHE efficiency varies accordingly. The ideal efficiency of the system is shown in Fig. 7. Higher temperature differentials lead to higher efficiencies. Also, in Fig. 7, if the temperature difference is constant, at lower temperatures, the efficiency is higher.

In the MHE assembly, the fluid mixture is 50% water and 50% glycol. This system can operate at temperatures below 0 °C. Increasing the concentration of the glycol in the mixture reduces the freezing point of the liquid. Hence, the MHE can operate at lower temperatures. Increasing the ratio of the glycol in the water/glycol mixture lowers the specific heat capacity (Fig. 8). To transfer a certain magnitude of heat through the heat exchangers, higher mass flow rates through the tube bundles are required. In this case, the energy usage by the MHE pumping system increases and the net power output from the MHE decreases. Also, addition of glycol increases the dynamic viscosity of the mixture. As the temperature decreases, the kinematic viscosity increases with temperature (see Fig. 8).

Since the heat sink of the MHE in a cement plant is at ambient conditions, the atmospheric temperature has a significant effect on the heat engine efficiency. In Fig. 9, this variation is shown. The measured average mechanical efficiency of the first MHE prototype is about 40% [22]. As listed in Table 2, about 20% of the energy input to the piston assembly is lost in the form of friction. The other 20% of losses occurs in the chains, belt and electricity generator. The piston assembly has an average of 60 strokes per minute. This results in rotating the flywheel at an average speed of 44.6 RPM [22]. In addition to the mechanical losses, about a 50% loss occurs in the heat exchangers and pipes [15]. An atmospheric temperature of

Table 2

Torque output of the piston assembly with the diameter of 63.5 mm.

|                            |       |       |       |        |
|----------------------------|-------|-------|-------|--------|
| Pressure difference (kpa)  | 344.7 | 517.1 | 689.5 | 1723.7 |
| Ideal torque output (N.m)  | 24.3  | 36.4  | 48.6  | 121.4  |
| Actual torque output (N.m) | 19.4  | 29.1  | 38.9  | 97.1   |

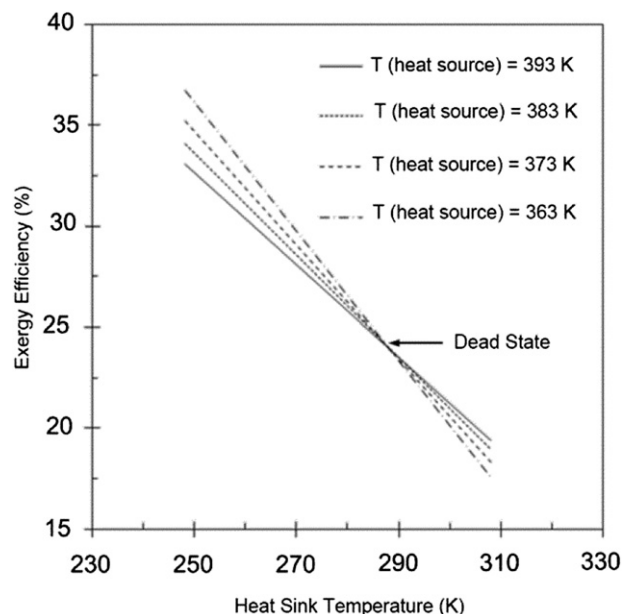


Fig. 10. Variation of MHE exergy efficiency with varying heat sink temperatures.

288 K is assumed. Fig. 9 shows how the heat recovered from the system varies when the production capacity changes. The maximum power from the MHE generator is 110 kW. This occurs when the cement plant is producing about 5400 tonnes of clinker per day.

An exergy analysis is performed to assess the efficiency of the system operation. Exergy lost and destroyed within each component will characterize how internal and external irreversibilities diminish the performance of the MHE. The exergy recovered in the MHE can be related to reduced electricity consumption by the system. The available exergy input is the exergy associated with the hot stream that flows into the heat exchangers. Fig. 10 shows how the exergy efficiency of the system varies when the heat sink temperature decreases. Due to irreversibilities and heat losses, the maximum exergy efficiency of the MHE system is calculated as 36 percent. When the temperature of the heat sink is the dead state (288 K), all of the exergy efficiencies are equal. At higher ambient temperatures, the exergy efficiency is lower. Below the dead state temperature, reducing the heat source temperature further results in a lower exergy efficiency. Also, the exergy efficiency is higher than the energy efficiency. When low-grade heat is converted to electricity, it yields a higher grade form of energy. From the analysis of the exergy loss, the insulation of the pipe and heat exchanger shell have significant effects on the MHE efficiency. When the ambient temperature is significantly different than the heat source temperature, the amount of heat and exergy losses to the ambient increases.

#### 5. Conclusions

This paper has presented the promising performance of a new type of low temperature heat engine, called a Marnoch Heat Engine (MHE). New data was presented to show how the heat recovery system would perform in different ambient conditions. Higher temperature differentials result in higher power output from the heat engine. The efficiency of the MHE is higher on colder days, as the outlet gas temperature from the quenching system and the main stack are nearly constant. Also, the behaviour of water/glycol mixtures in different conditions in the coils was reported. When the glycol concentration increases in the liquid mixture, more power is consumed by the pumps to transport liquid through the

coils and valves. It was also shown that the mechanical configuration of the MHE has significant effects on the energy and exergy efficiency of the heat engine. The analysis results show that MHE is capable of generating electricity from waste heat in cement plants.

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